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NACA

RESEARCH MEMORANDUM

HEAT TRANSFER FROM HIGH-TEMPERATURE SURFACES TO FLUIDS

I - PRELIMINARY INVESTIGATION WITH AIR IN INCONEL TUBE WITH ROUNDED ENTRANCE, INSIDE DIAMETER OF 0.4 INCH AND LENGTH OF 24 INCHES

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NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS

WASHINGTON

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of 0.402 inch, a wall thickness of 0.049 inch, and a total length of 24.75 inches. Steel flanges welded to the tube near each end provide electrical contact with heavy brass flanges, which are in turn connected by copper cables to the electrical power source. The distance between the outer faces of the steel tube flanges (24 in.) is taken as the effective heat-transfer length. The heater tube entrance consists of an A.S.M.E. long-radius flow nozzle, which has the same throat diameter as the inside diameter of the tube. This nozzle serves as a smooth entrance to the tube and is also used to measure the air flow.

A static-pressure tap at the throat of the entrance nozzle is used to measure the tube entrance static pressure. Additional static-pressure taps are located at intervals along the tube (fig. 2).

The heater tube is thermally insulated by a covering of insulating cement, which is surrounded by three concentric stainless-steel radiation shields with insulating cement in the spaces between the shields. The total thickness of the insulation and shielding is 2.5 inches.

Outside wall temperatures of the tube are measured at 30 locations (two thermocouples located 180° apart at each of 15 stations) by means of chromel-alumel thermocouples (26-gage flexible glass insulated wire) and a self-balancing indicating-type potentiometer. The thermocouples were attached to the tube by peening the wires into holes having a nominal depth of 1/32 inch; the wire junctions were located in planes normal to the tube axis. Care was taken that the last point of contact between the wires was at the outside surface of the tube wall. The thermocouples were wrapped around approximately one-fourth of the circumference of the tube in order to reduce conduction losses.

A photograph of the heater tube installed in the setup is shown in figure 3.

Air System

Air at a pressure of about 100 pounds per square inch gage is supplied through a pressure-regulating valve, cleaner, and preheater to a large surge tank (fig. 1). From the surge tank, the air passes through a second pressure-regulating valve and a rotameter into a calming tank and then through the heater tube into a mixing tank, from which it is discharged to the atmosphere. The surge tank and the tanks at the entrance and exit of the heater tube are thermally insulated.

Air pressures from 0 to 65 pounds per square inch gage and temperatures from 80° to 400° F are available at the entrance to the heater tube.

The mixing tank at the heater-tube exit is made up of three concentric passages so arranged that the hot air leaving the tube makes three passes axially through the tank before being discharged. Baffles are provided in the central passage to insure thorough mixing of the air leaving the heater tube.

The temperature of the air entering the heater tube is measured by three iron-constantan thermocouples located in the calming tank.

The temperature of the air leaving the tube is measured by two chromel-alumel thermocouples located downstream of the baffles in the mixing tank.

A stagnation-type Franz thermocouple probe, having a calibrated recovery factor of about 0.94 and arranged to traverse the tube center line, is provided for estimating the air-temperature distribution along the tube. The thermocouple is mounted on a support that passes through the downstream end of the mixing tank on the center line and enters the exit end of the heater tube. The probe is completely withdrawn from the tube during a test run except when the air-temperature distribution is being obtained.

Electrical System

Power is supplied to the heater tube from a 208-volt, 60-cycle supply line through an autotransformer and a 20:1 power transformer. The low voltage leads of the power transformer are connected to the heater tube by copper cables, as previously described.

The electrical power input to the heater tube is measured by a wattmeter. In addition, an ammeter is used to measure the current through the tube and a voltmeter connected across the tube at the brass connector flanges is used to measure the voltage drop. The ammeter and current coil of the wattmeter are connected to the load through a 240:1 instrument current transformer.

The capacity of the electrical equipment is 15 kilovolt-amperes.

PROCEDURE

Calibration Tests

A number of preliminary investigations were made in order to check the instrumentation.

Air-flow measurements. - The flow nozzle at the heater-tube entrance was calibrated for the unheated condition by using the large surge tank as an air reservoir. Air was allowed to flow from the surge tank through the nozzle; the pressure at the nozzle entrance was held constant. The pressure and temperature in the surge tank were recorded at selected time intervals; by use of the equation of state, the flow rate was obtained. The flow rates so obtained were from 0 to 2 percent higher than the values calculated from A.S.M.E. equations. The rotameter, which had been calibrated by a gasometer, showed about the same deviation from the calculated flows.

Tube-wall-temperature measurements. - In order to determine whether errors existed in tube-wall-temperature measurements due to the current in the heater tube, power was supplied to the tube with no air flowing until a predetermined wall temperature, as indicated by the thermocouples, was obtained. The power was then shut off and the readings of several thermocouples were recorded at 5-second intervals. The temperatures determined by extrapolation to zero time of the temperature against time curve thus obtained for each thermocouple agreed with the temperatures read just prior to the power shutoff; hence it was concluded that the current had no effect on wall-temperature measurements.

Heat losses. - The heat loss from the tube was obtained for the range of tube-wall temperature by supplying various amounts of power to the heater tube with no air flowing through the system. After equilibrium conditions had been maintained for approximately 1/2 hour, the power input and all tube-wall temperatures were recorded. The power input for a given average wall temperature with no flow was considered to be the heat loss for the same average wall temperature with air flowing in the tube.

Heat-Transfer Tests

Investigations were conducted to obtain surface heat-transfer coefficients and the associated static-pressure drops for average inside-tube-wall temperatures from 220° to 1240° F (corresponding

maximum local wall temperatures, 250° to 1500° F), Reynolds numbers from 10,000 to 250,000, and the tube-exit Mach numbers up to 1.0. The air temperature at the tube entrance was about 85° F and the pressure varied from 15 to 45 pounds per square inch absolute.

The experimental procedure was as follows: The entrance-air pressure in the calming tank was set at the minimum value and power was supplied to the heater tube until the desired tube-wall temperature was obtained. After equilibrium conditions had been maintained for approximately 1/2 hour, all power input, pressure, and temperature readings were recorded with the Franz thermocouple probe retracted in the outlet mixing tank. The probe was then moved upstream along the center line of the tube and the air temperature was recorded at 4-inch intervals. This procedure was repeated over the range of entrance pressure for five wall-temperature levels.

SYMBOLS

The following symbols are used in the calculations:

c_p	specific heat of air at constant pressure, (Btu/(lb)(°F))
D	inside diameter of heater tube, (ft)
G	mass velocity of air, (lb/(hr)(sq ft))
h	heat-transfer coefficient, (Btu/(hr)(sq ft)(°F))
k	thermal conductivity of air, (Btu/(hr)(sq ft)(°F/ft))
k_I	thermal conductivity of Inconel, (Btu/(hr)(sq ft)(°F/ft))
L	effective heat-transfer length of heater tube, (ft)
Δp	static-pressure drop across heater tube, (lb/sq ft)
Q	rate of heat transfer to air, (Btu/hr)
R_i	inner radius of heater tube, (ft)
R_o	outer radius of heater tube, (ft)
S	heat-transfer area of heater tube, (0.211 sq ft)
T_1	total air temperature at entrance to heater tube, (°F)

T_2	total air temperature at exit of heater tube, (°F)
T_b	average bulk temperature of air, $\frac{1}{2}(T_1 + T_2)$, (°F)
T_s	average inside-wall temperature of heater tube, (°F)
T_o	average outside-wall temperature of heater tube, (°F)
V_b	bulk velocity of air, (ft/hr)
W	air flow, (lb/hr)
μ	absolute viscosity of air, (lb/(ft)(hr))
ρ	density of air, (lb/cu ft)
ρ_1/ρ_2	ratio of densities at entrance and exit of heater tube
σ_1	ratio of density at tube entrance to standard sea-level density, $\rho_1/0.0765$
$c_p\mu/k$	Prandtl number
hD/k	Nusselt number
DG/μ	Reynolds number
$\rho_s V_b D/\mu_s$	modified Reynolds number

Subscripts:

b	physical properties of air evaluated at average bulk temperature (average total temperature T_b)
s	physical properties of air evaluated at average inside-wall temperature T_s

METHODS OF CALCULATION

Temperatures. - The average outside-tube-wall temperature T_o was obtained by plotting curves of temperature against tube length, measuring the area under the curve, and dividing by the tube length. The plotted values of temperature were the average of the two thermocouples located 180° apart at each of the 15 stations along the tube.

The average inside-tube-wall temperature T_s was calculated from the outside-tube-wall temperature, the heat flow, and thermal conductivity and physical dimensions of the tube by the following equation, which can be derived (reference 1) with the assumptions that heat is uniformly generated across the tube wall and that heat flow is directed radially inward:

$$T_s = T_o - \frac{Q}{2\pi L k_I (R_o^2 - R_i^2)} \left(R_o^2 \log_e \frac{R_o}{R_i} - \frac{R_o^2 - R_i^2}{2} \right)$$

When the dimensions of the heater tube and a value of 8.6 for the thermal conductivity of Inconel are substituted in this equation

$$T_s = T_o - 0.00107\ Q$$

A constant value of thermal conductivity was used because of lack of data on variation with temperature. The error introduced is negligible inasmuch as the difference between inside- and outside-wall temperature is small compared with the difference between average wall and air temperature (for example, maximum value - 23° F temperature difference between outer and inner wall at 940° F temperature difference between wall and air).

The average bulk air temperature T_b was taken as the arithmetic mean of the total temperatures T_1 and T_2 of the air measured in the inlet and the outlet tanks.

Heat-transfer coefficients. - Average heat-transfer coefficients h were calculated from the following equation:

$$h = \frac{W c_{p,b} (T_2 - T_1)}{S \left(T_s - \frac{T_2 + T_1}{2} \right)} = \frac{W c_{p,b} (T_2 - T_1)}{S (T_s - T_b)}$$

The use of average air temperature in computing the heat-transfer coefficient is considered satisfactory inasmuch as the temperature rise of the tube wall along its length, neglecting the end points, was about equal to the rise in air temperature from inlet to outlet, as will be shown later.

Physical properties of air. - The specific heat at constant pressure c_p, the thermal conductivity k, the viscosity μ, and the corresponding Prandtl number $c_p\mu/k$ used were taken from reference 2. These physical properties are plotted in figure 4 as functions of temperature.

RESULTS AND DISCUSSION

Temperature Distribution

The axial distribution of outside-tube-wall temperatures along the tube length for the condition of no air flow is shown in figure 5 for various amounts of electrical heat input. High temperature gradients exist at the entrance and the exit of the tube as a result of end conduction losses through the electrical connector flanges and cables; the temperature, however, is reasonably constant along the greatest portion of the tube length. The slight dips, which occur in the curves between 8 and 12 inches from the tube entrance, are not understood at this time.

Representative axial distributions of outside-tube-wall temperatures are shown in figure 6 for five different amounts of total electrical heat input at an approximately constant average Reynolds number of 140,000. The Reynolds number was computed at the average air temperature T_b, hence the air flow had to be increased with increasing heat input and attendant increasing exit and average air temperature in order to maintain constant Reynolds number. The temperature increases almost linearly along the central 20 inches of the tube and, similar to the no-flow case (fig. 5), decreases sharply at both ends of the tube.

The distribution of unmixed air temperature along the center line of the tube, as indicated by the Franz thermocouple probe, is shown in figure 7 for the same conditions as figure 6. Included in the figure at the 24-inch station are the mixed exit-air temperatures as measured in the mixing tank. The unmixed air temperature increases almost linearly along the central 16 to 20 inches of tube length and levels off at both ends because of decreased tube-wall temperature (fig. 6). The mixed exit-air temperature is higher than the unmixed air temperature indicated by the probe at the 24-inch station; the difference in temperature increases from 17° F at the lowest heat input to 180° F at the highest heat input. These differences in temperature might be explained by the existence of large temperature gradients across the air stream in the tube.

Examination of figures 6 and 7 indicates that the rise in mixed air temperature (fig. 7) is approximately equal to the increase in wall temperature along the tube length if end effects are neglected (fig. 6).

Heat Balance

The external heat losses from the heater tube as obtained from figure 5 are cross-plotted against average outside-wall temperature of the heater tube in figure 8. The average outside-tube-wall temperatures were obtained by integration of the curves of figure 5. A heat balance for all the heat-transfer experiments is shown in figure 9 in which values of the heat transferred to the air $\left[Wc_{p,b}(T_2 - T_1)\right]$ plus the heat loss (obtained from fig. 8) are plotted against electrical heat input.

A good heat balance is indicated with a maximum deviation of about 5 percent from the match line.

Correlation of Heat-Transfer Coefficients

Correlation based on bulk temperatures. - Forced convection, turbulent flow, heat-transfer coefficients for low viscosity fluids are generally correlated by means of the familiar Nusselt relation (reference 3) where Nusselt number divided by Prandtl number to a power (generally 0.4) $\left(\frac{hD}{k_b}\right)\Big/\left(\frac{c_{p,b}\,\mu_b}{k_b}\right)^{0.4}$ is plotted against Reynolds number (DG/μ_b) and in which the physical properties of the fluid are evaluated at the average fluid bulk temperature. The results of the present investigation are plotted in this manner in figure 10. Included for comparison is the average line (dashed) obtained in reference 3 from a correlation of the results of various investigators and the average line (dotted) corresponding to the correlation obtained in reference 4 for the data of the same investigators using the same physical properties (reference 2) used herein (fig. 4). The dotted line was obtained by transposition of the Stanton number plot of reference 4 to the coordinates of figure 10 wherein Prandtl number was assigned a value of 0.70 corresponding to the physical properties of air (fig. 4) at a temperature of 120° F. The line corresponding to the correlation of reference 4, although obtained for the same data, is

lower than that of reference 3, undoubtedly because of use of different physical properties. The lower line is considered preferable for comparison with the correlation of the data of this investigation inasmuch as the same physical properties are used.

A family of parallel lines for the different temperature levels is obtained for the present data with slopes of approximately 0.8 at Reynolds numbers above approximately 25,000. At lower Reynolds numbers the data fall off indicating the presence of transition. The low temperature data (220° F wall temperature) are in good agreement with the line of reference 4; however, the higher temperature-data lines decrease progressively with increased wall temperature and corresponding increased temperature difference between wall and air. The values of $\left(\frac{hD}{k_b}\right)\Big/\left(\frac{c_{p,b}\,\mu_b}{k_b}\right)^{0.4}$ for constant Reynolds number decrease about 25 percent with increase in average wall temperature from 220° to 1240° F.

Correlation based on surface temperature. - The data of figure 10 are replotted in figure 11 wherein the physical properties are evaluated at the inside-tube-wall or surface temperature instead of the air bulk temperature. This plot shows a similar separation of the data with temperature level as obtained in figure 10 except that the separation is increased; the values of $\left(\frac{hD}{k_s}\right)\Big/\left(\frac{c_{p,s}\,\mu_s}{k_s}\right)^{0.4}$ decrease approximately 40 percent as the tube temperature level increases from 220° to 1240° F. If the physical properties were evaluated at the film temperature (average of the surface and bulk temperatures), the separation of the data with temperature level would be between the 25 and 40 percent indicated in figures 10 and 11, respectively.

Correlation based on modified Reynolds number. - A plot of the data is shown in figure 12 wherein the physical properties are again evaluated at the surface temperature but in addition Reynolds number is modified by substituting the product of air density evaluated at the surface temperature ρ_s and average air velocity evaluated at the air bulk temperature V_b for the conventional mass velocity G or $\rho_b V_b$. In addition, the viscosity term in Reynolds number is evaluated at the surface temperature. The relation between the modified and the conventional Reynolds number is given by the following equation:

$$\frac{\rho_s V_b D}{\mu_s} = \left(\frac{DG}{\mu_s}\right)\left(\frac{\rho_s}{\rho_b}\right) = \left(\frac{DG}{\mu_s}\right)\left(\frac{T_b + 460}{T_s + 460}\right) = \left(\frac{DG}{\mu_b}\right)\left(\frac{T_b + 460}{T_s + 460}\right)\left(\frac{\mu_b}{\mu_s}\right)$$

This definition of Reynolds number is essentially similar to that theoretically shown in reference 5 to be the correct value for laminar heat transfer from a flat plate. This method of plotting (fig. 12) results in a good correlation of the data for Reynolds numbers above transition for all the temperature levels investigated (average wall temperatures up to 1240° F and corresponding maximum local wall temperatures up to 1500° F). The equation of the line, which represents the data for Reynolds numbers above approximately 10,000, is

$$\left(\frac{hD}{k_s}\right) \Big/ \left(\frac{c_{p,s}\,\mu_s}{k_s}\right)^{0.4} = 0.022 \left(\frac{\rho_s V_b D}{\mu_s}\right)^{0.8}$$

In figure 13 the data are replotted neglecting Prandtl number, that is, hD/k_s is plotted against $\rho_s V_b D/\mu_s$. A good correlation is again obtained indicating that the variation of Prandtl number with temperature (fig. 4) has little effect. The equation of the line best representing the data is

$$\frac{hD}{k_s} = 0.018 \left(\frac{\rho_s V_b D}{\mu_s}\right)^{0.8}$$

Pressure-Drop Correlation

The measured static-pressure drops across the heater tube are correlated in figure 14 where the product of pressure drop and entrance density (referred to standard sea-level density) divided by air flow to the 1.8 power $\frac{\sigma_1 \Delta p}{W^{1.8}}$ is plotted against the ratio of entrance-to-exit density ρ_1/ρ_2. The exponent of 1.8 on air flow was obtained from preliminary logarithmic plots of $\sigma_1 \Delta p$ against W for constant values of ρ_1/ρ_2. The air densities ρ_1 (and corresponding σ_1) and ρ_2 were calculated using static pressure and total temperature at entrance and exit of the tube, respectively. Total instead of static temperature was used in

order to simplify the calculations involved in reducing the experimental data. All the pressure-drop data correlate within a deviation of about ±1 percent except for four points, which were obtained at Reynolds numbers below 10,000. The equation of the line through the data is

$$\frac{\sigma_1 \Delta p}{W^{1.8}} = 0.10 \frac{\rho_1}{\rho_2} - 0.044$$

SUMMARY OF RESULTS

The results of the heat-transfer investigation conducted with air flowing through an electrically heated Inconel tube with a rounded entrance, an inside diameter of 0.402 inch, and a length of 24 inches over a range of Reynolds numbers up to 250,000 and a range of average inside-tube-wall temperatures up to 1240° F (corresponding maximum local tube-wall temperatures up to 1500° F) showed that:

1. Correlation of the average heat-transfer coefficients according to the familiar Nusselt relation wherein physical properties were evaluated at the average bulk temperature resulted in a separation with temperature level of the parameter Nusselt number divided by Prandtl number to the 0.4 power; at constant Reynolds number, the parameter decreased about 25 percent with increase in average surface temperature from 220° to 1240° F. Evaluation of the physical properties at the average surface temperature reduced the parameter by 40 percent for the same increase in temperature.

2. A good correlation of the heat-transfer data was obtained for the entire range of temperatures when the viscosity μ_s, thermal conductivity k_s, and the specific heat $c_{p,s}$ of the air were evaluated at the average surface temperature and the Reynolds number was modified by substituting the product of air density evaluated at the average surface temperature ρ_s and velocity evaluated at the average air bulk temperature V_b for the conventional mass velocity G. The equation of the line representing the data for Reynolds numbers above 10,000 is

$$\left(\frac{hD}{k_s}\right) \Big/ \left(\frac{c_{p,s}\,\mu_s}{k_s}\right)^{0.4} = 0.022\left(\frac{\rho_s\, V_b\, D}{\mu_s}\right)^{0.8}$$

Satisfactory correlation was also obtained neglecting Prandtl number in which case the equation for the line fitting the data is

$$\frac{hD}{k_s} = 0.018 \left(\frac{\rho_s V_b D}{\mu_s}\right)^{0.8}$$

3. The pressure-drop data were correlated by plotting the product of static-pressure drop Δp and entrance air density (referred to standard sea-level density) σ_1 divided by air flow W raised to an exponent against the ratio of entrance-to-exit density ρ_1/ρ_2. The equation representing the data within a ±1 percent deviation for Reynolds numbers over 10,000 is

$$\frac{\sigma_1 \Delta p}{W^{1.8}} = 0.10 \frac{\rho_1}{\rho_2} - 0.044$$

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Cleveland, Ohio.

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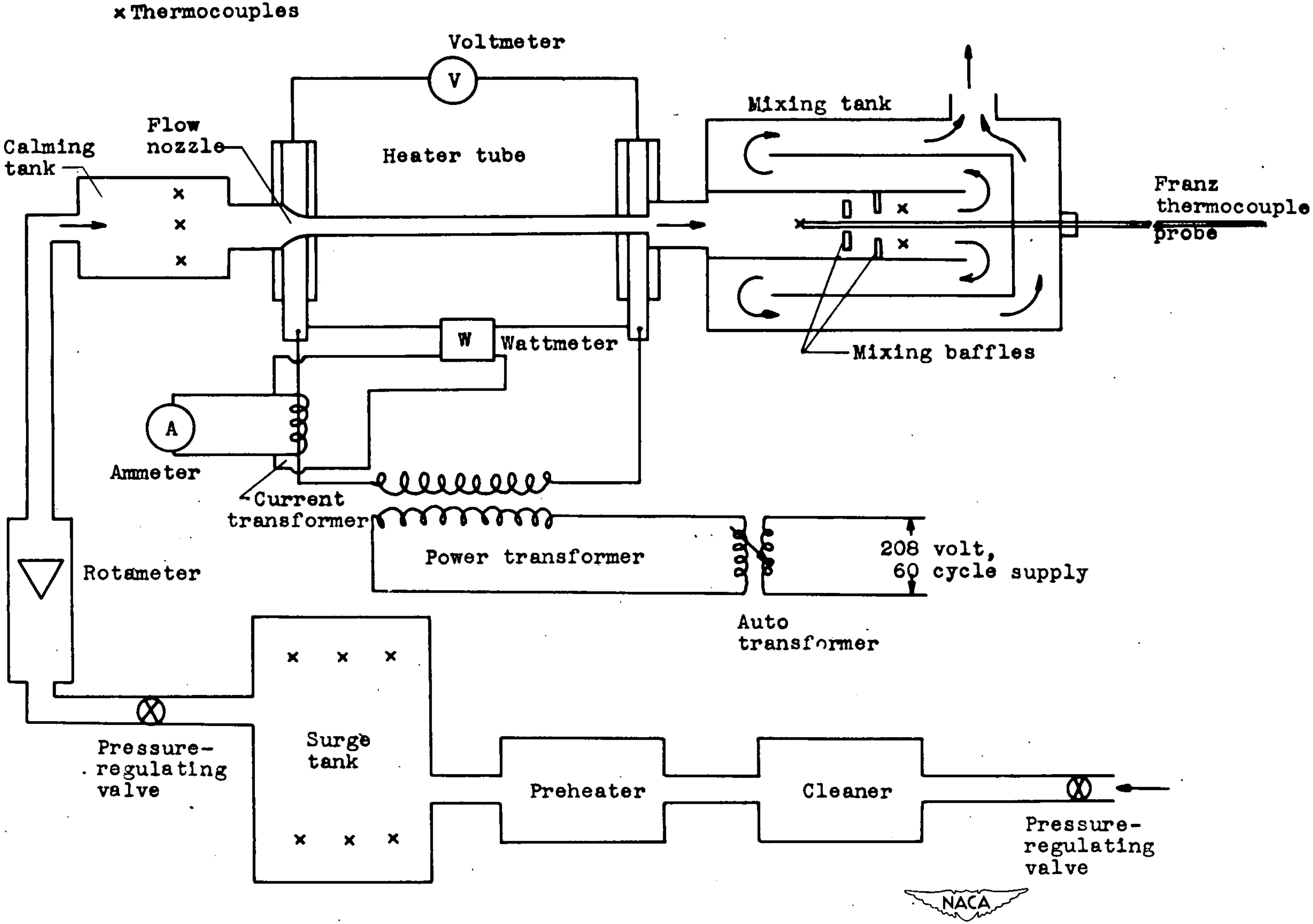

Figure 1. - Schematic diagram showing arrangement of experimental apparatus.

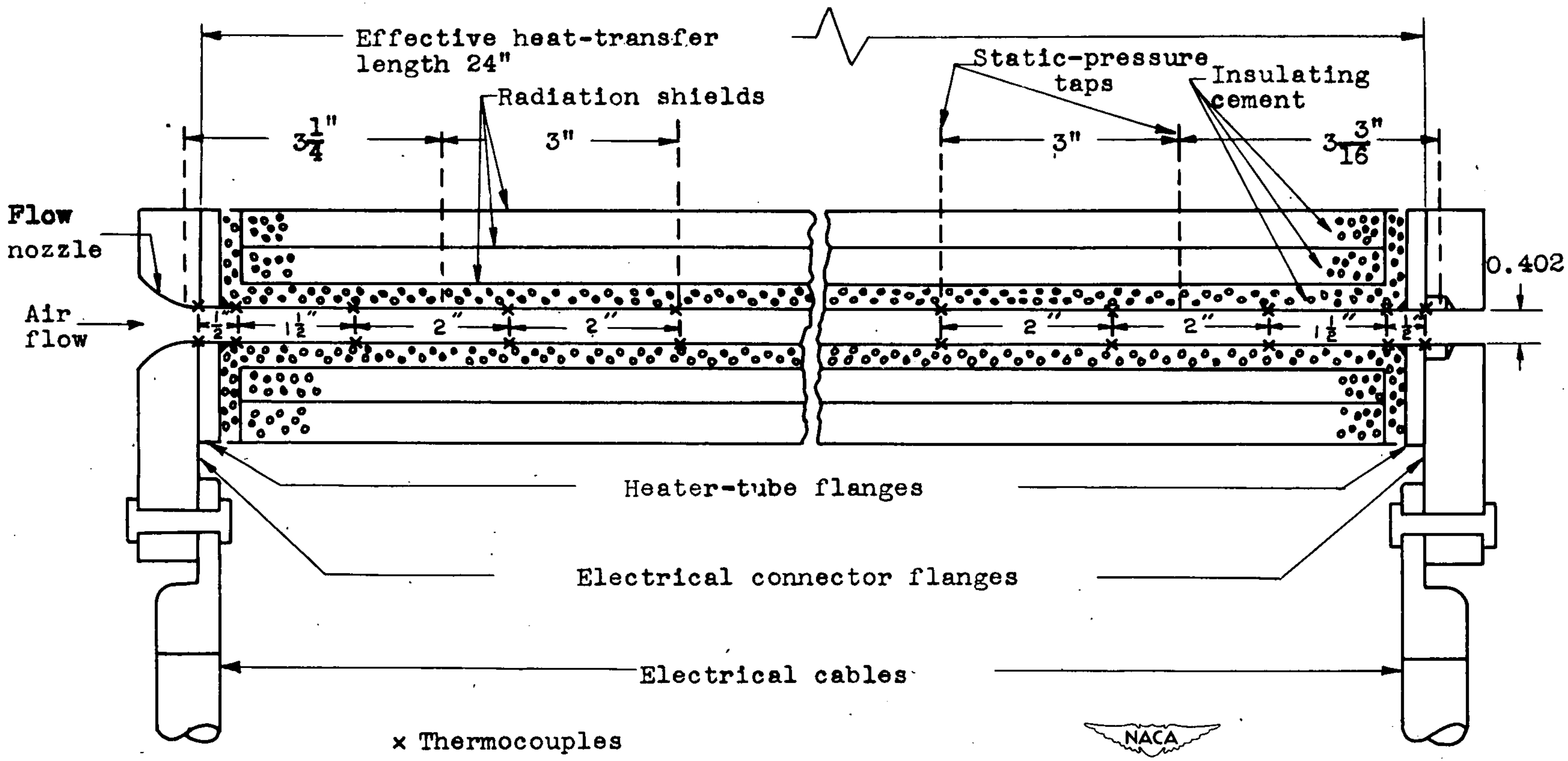

Figure 2. - Schematic diagram of test section showing thermocouple and pressure-tap locations.

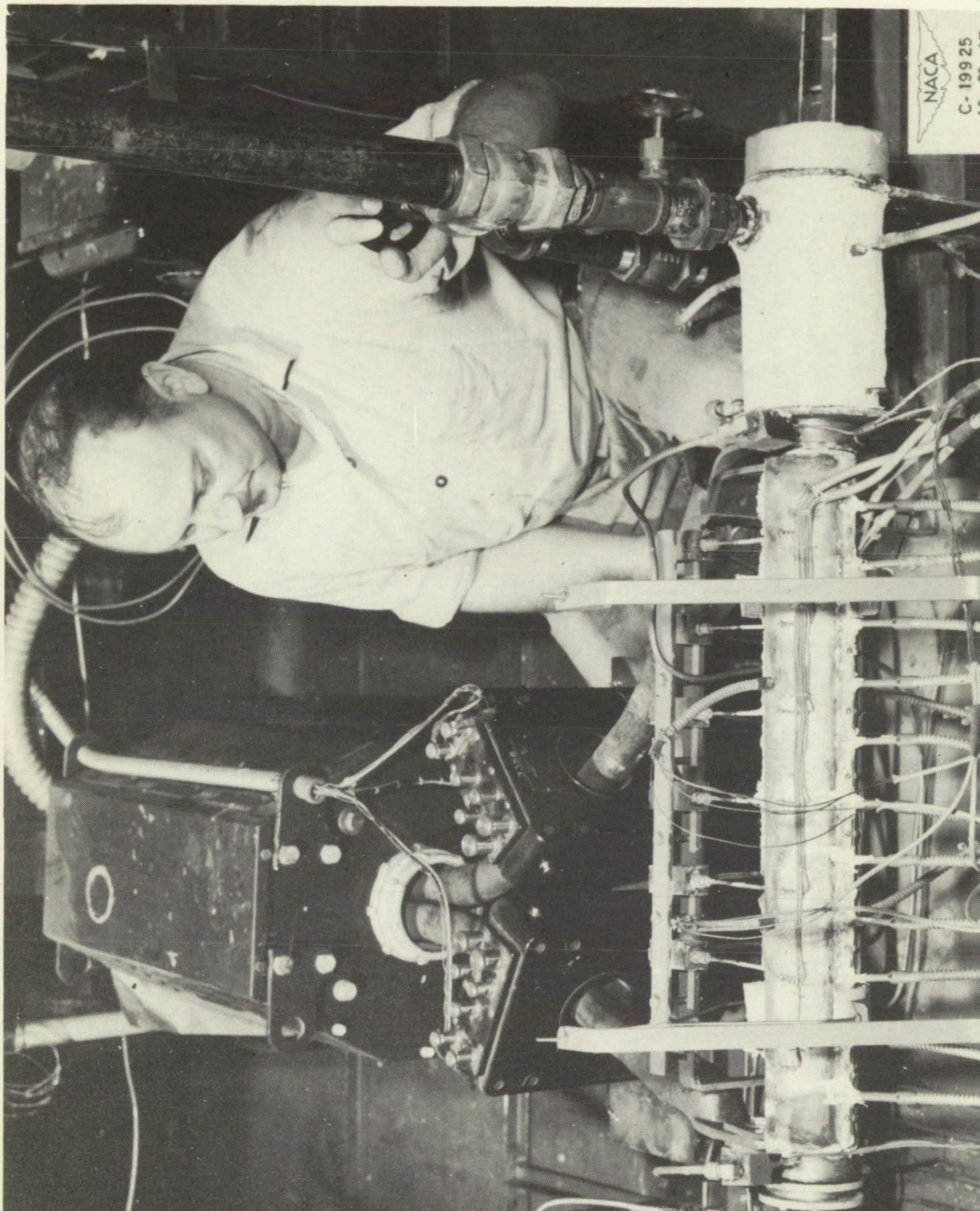

Figure 3. - General view of heater-tube installation.

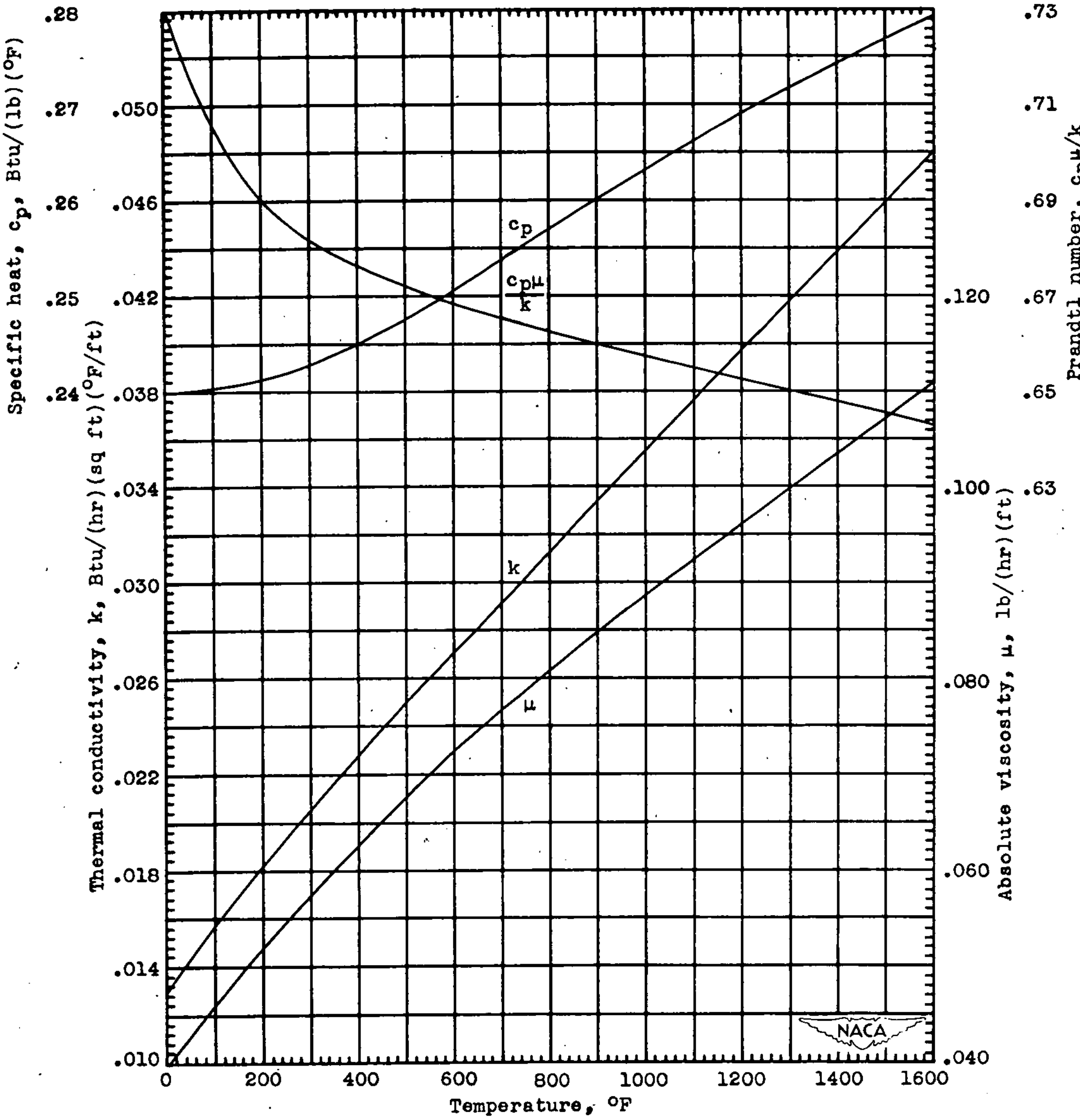

Figure 4. - Variation of specific heat c_p, thermal conductivity k, absolute viscosity μ, and Prandtl number $c_p\mu/k$ of air with temperature. (Data from reference 2.)

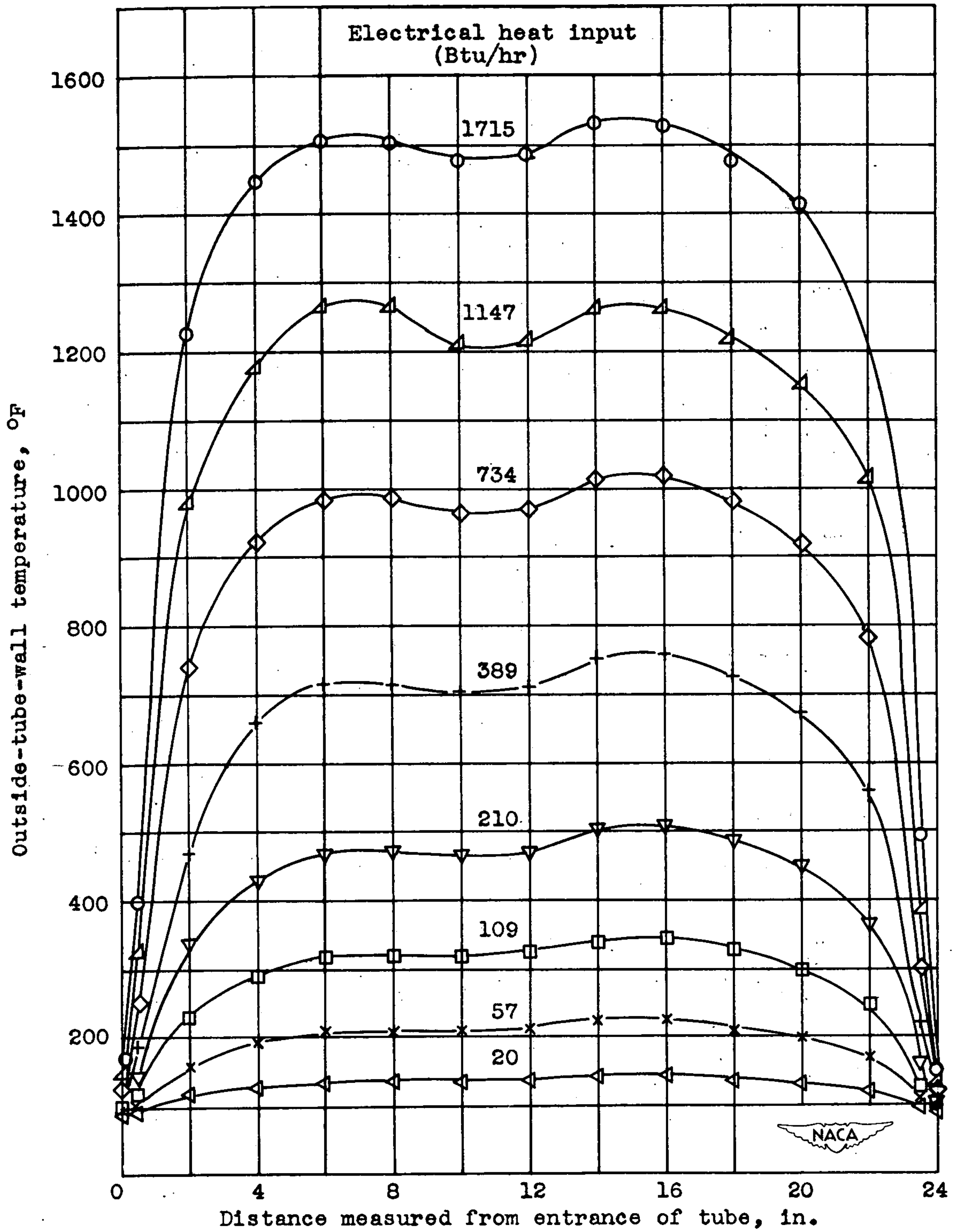

Figure 5. - Axial distribution of outside-tube-wall temperature for various amounts of electrical heat input with no air flow.

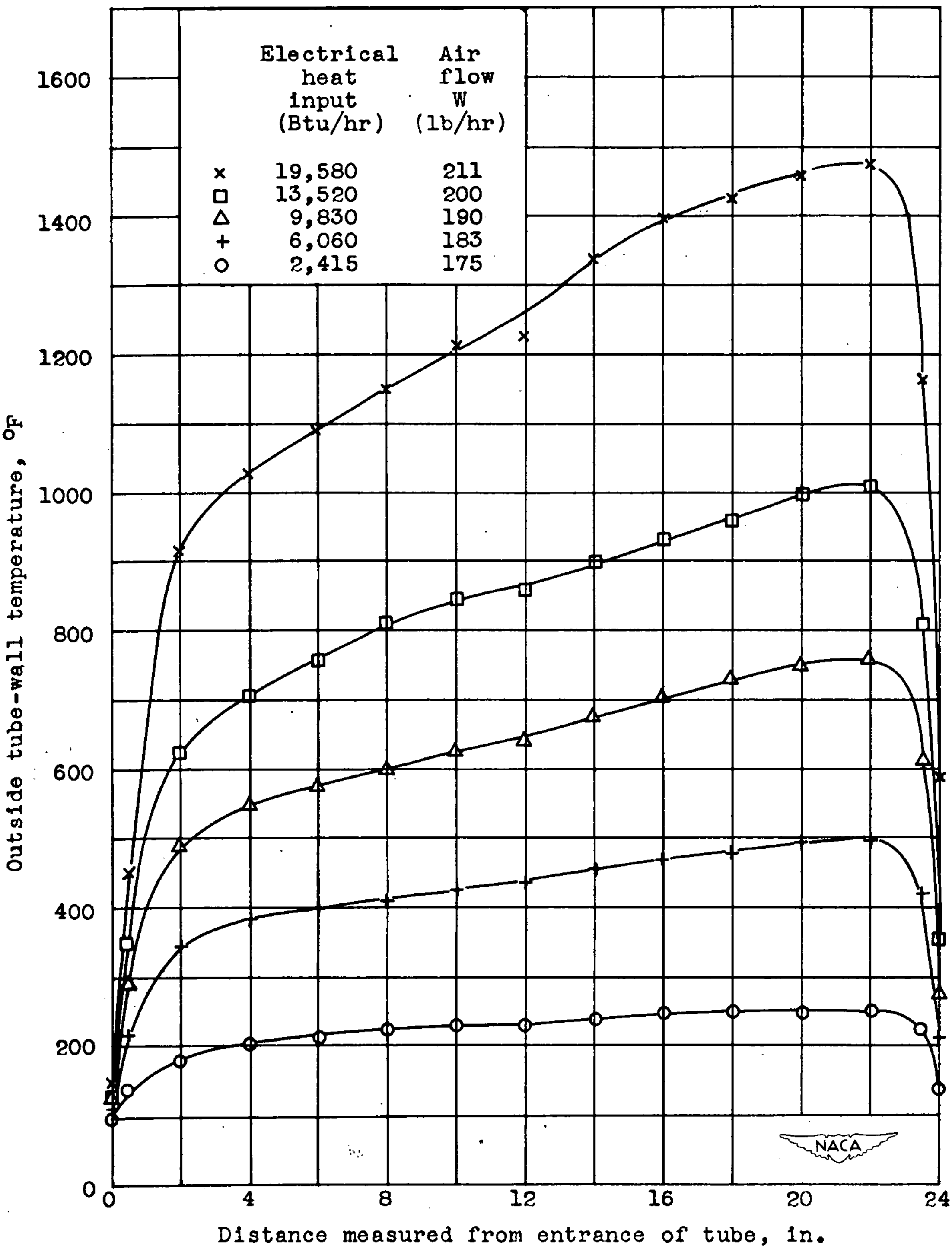

Figure 6. - Representative outside-tube-wall temperature distributions for various amounts of electrical heat input and a constant Reynolds number of approximately 140,000.

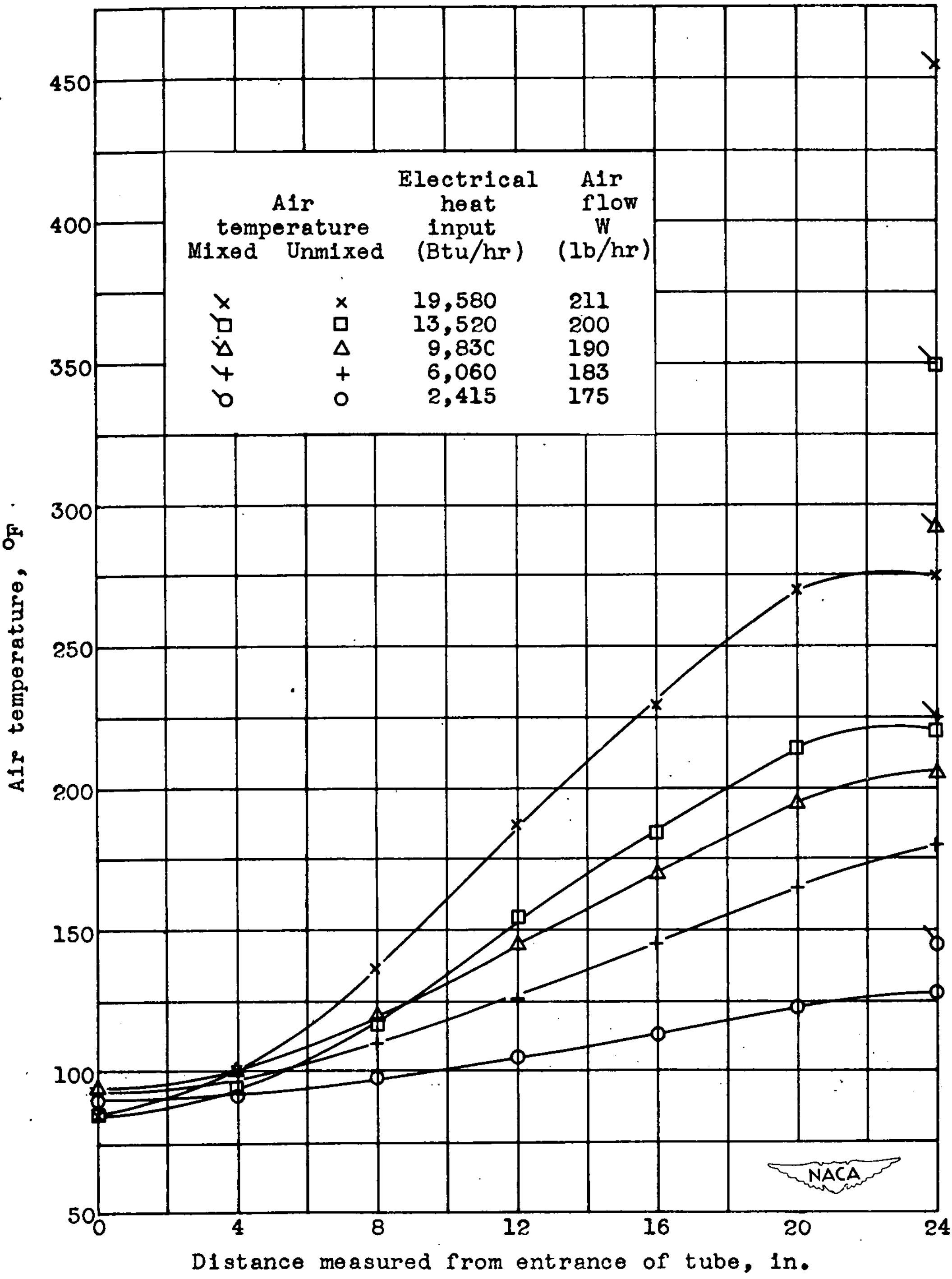

Figure 7. - Variation of the unmixed air temperature along the center line of the tube and the exit mixture temperature for various amounts of electrical heat input and an approximately constant Reynolds number of 140,000.

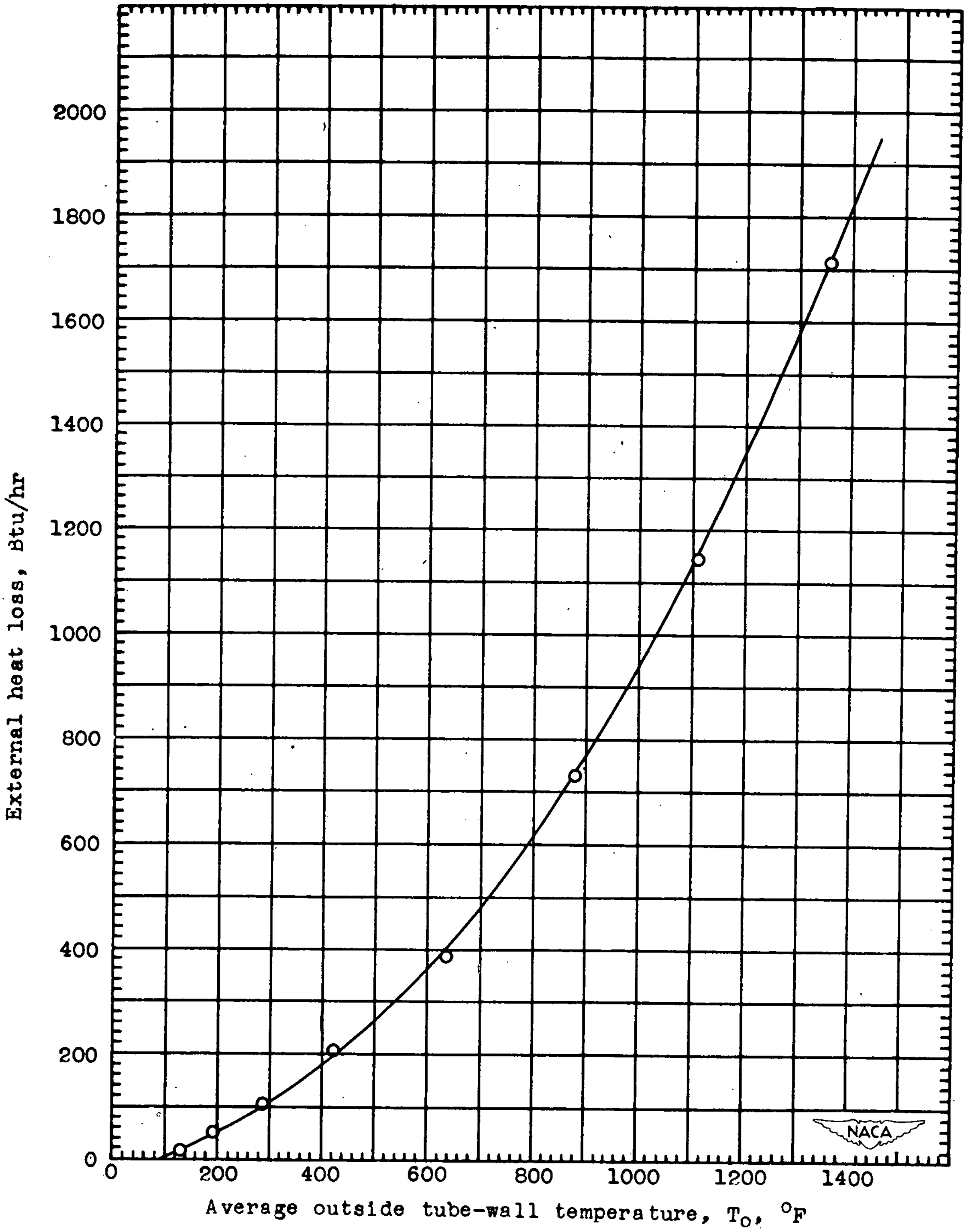

Figure 8. - Variation of external heat loss with average outside-tube-wall temperature.

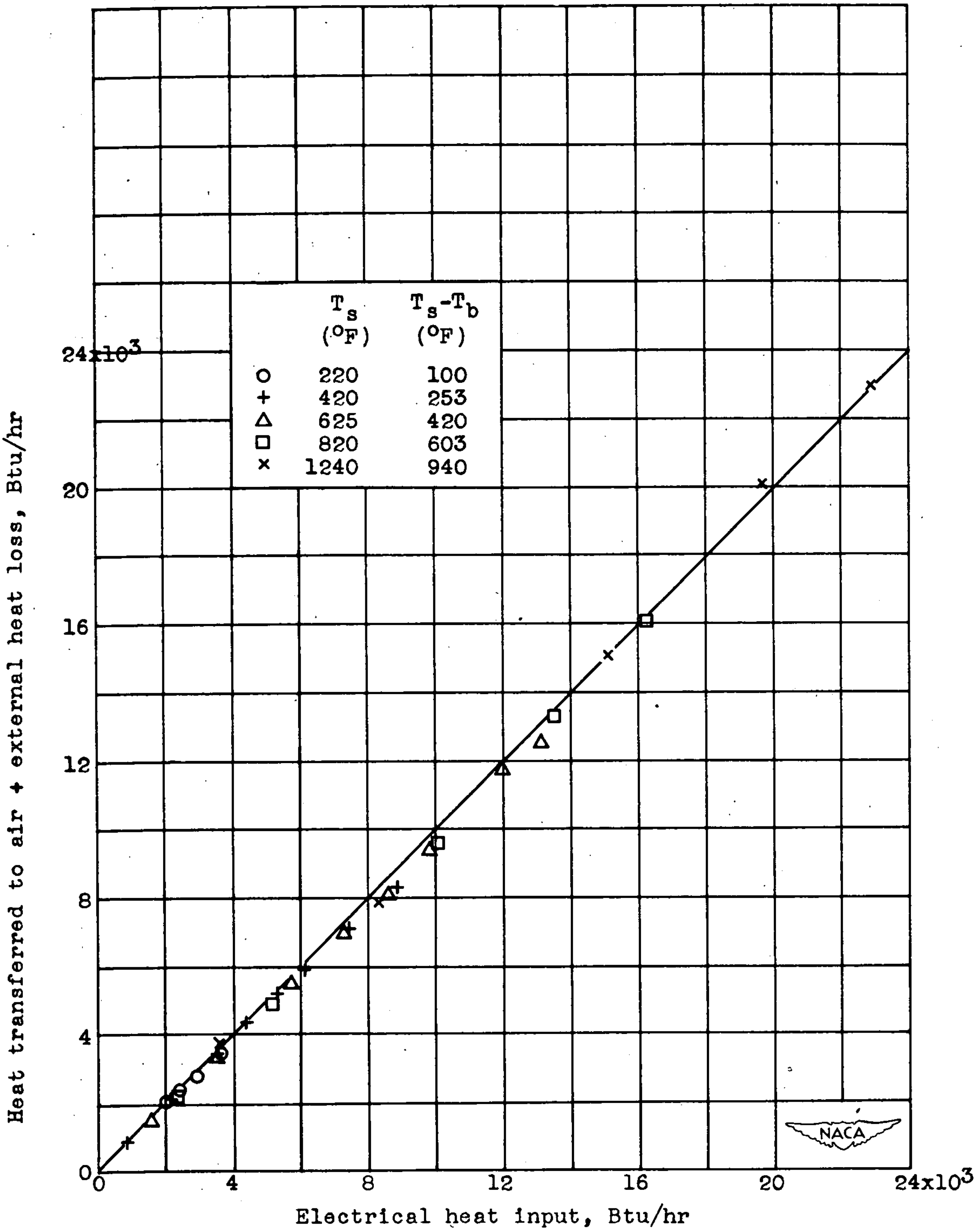

Figure 9. - Heat balance.

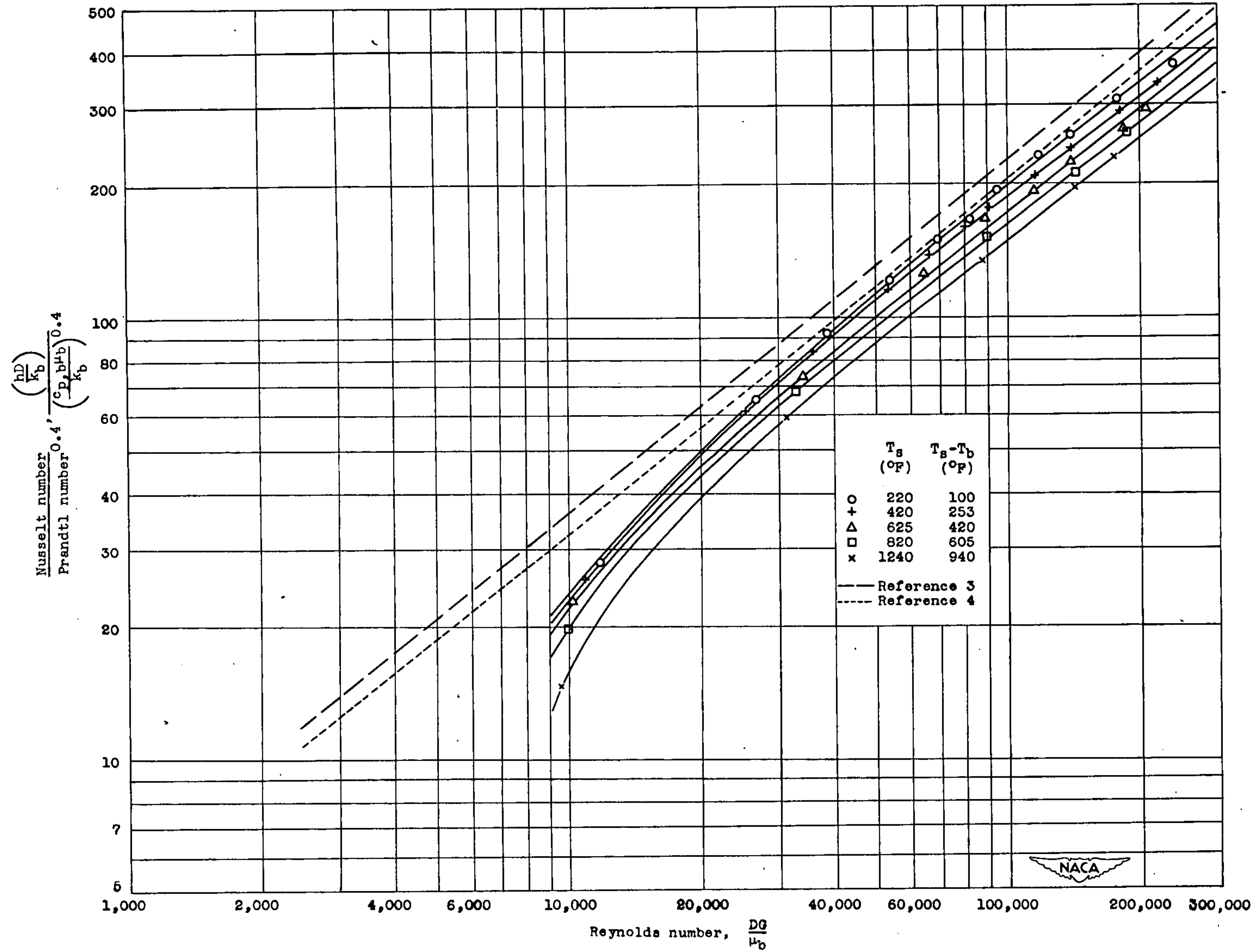

Figure 10. - Correlation of heat-transfer coefficients. Physical properties of air evaluated at the average bulk temperature T_b. To.

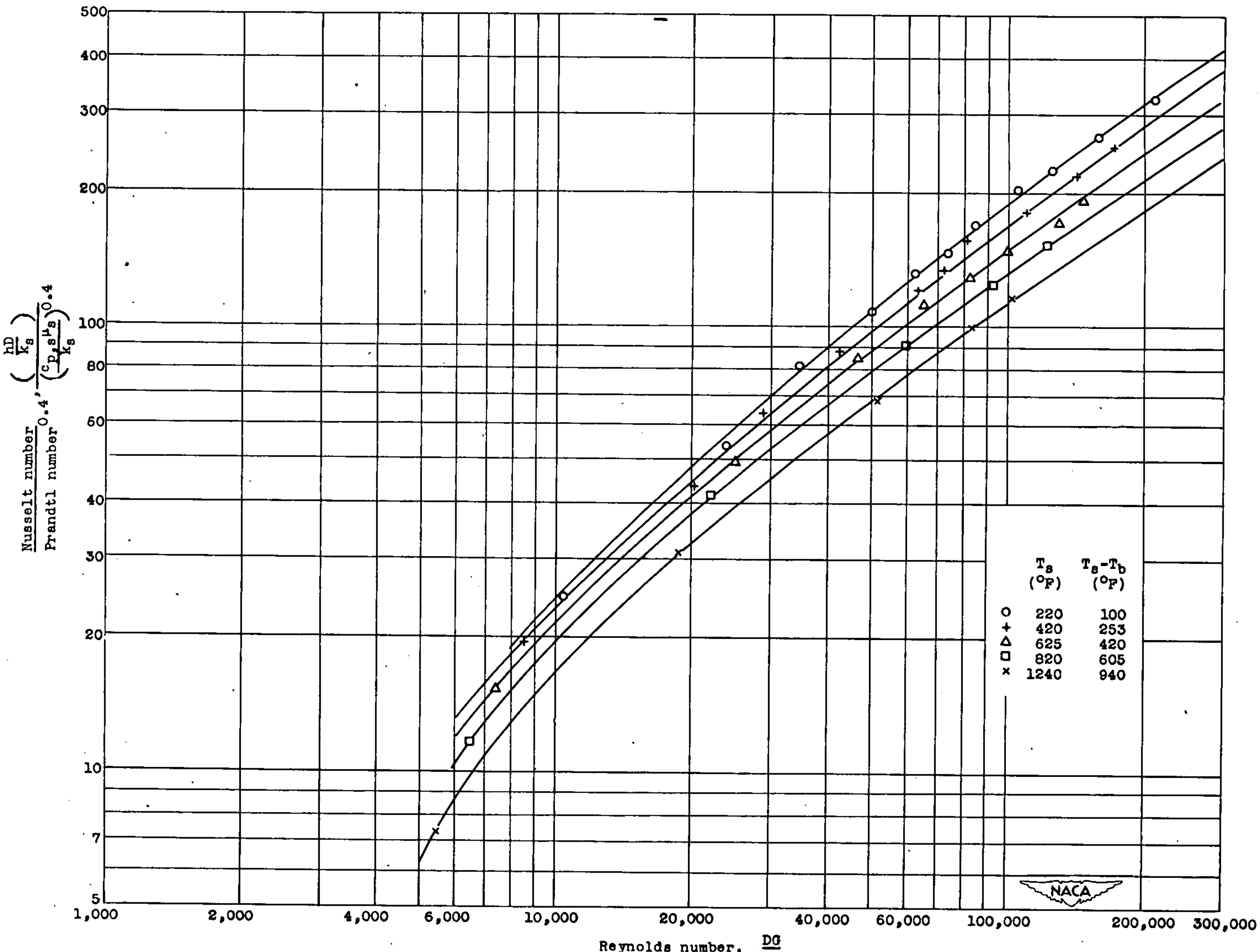

Figure 11. - Correlation of heat-transfer coefficients. Physical properties of air evaluated at the average inside-tube-wall temperature T_s.

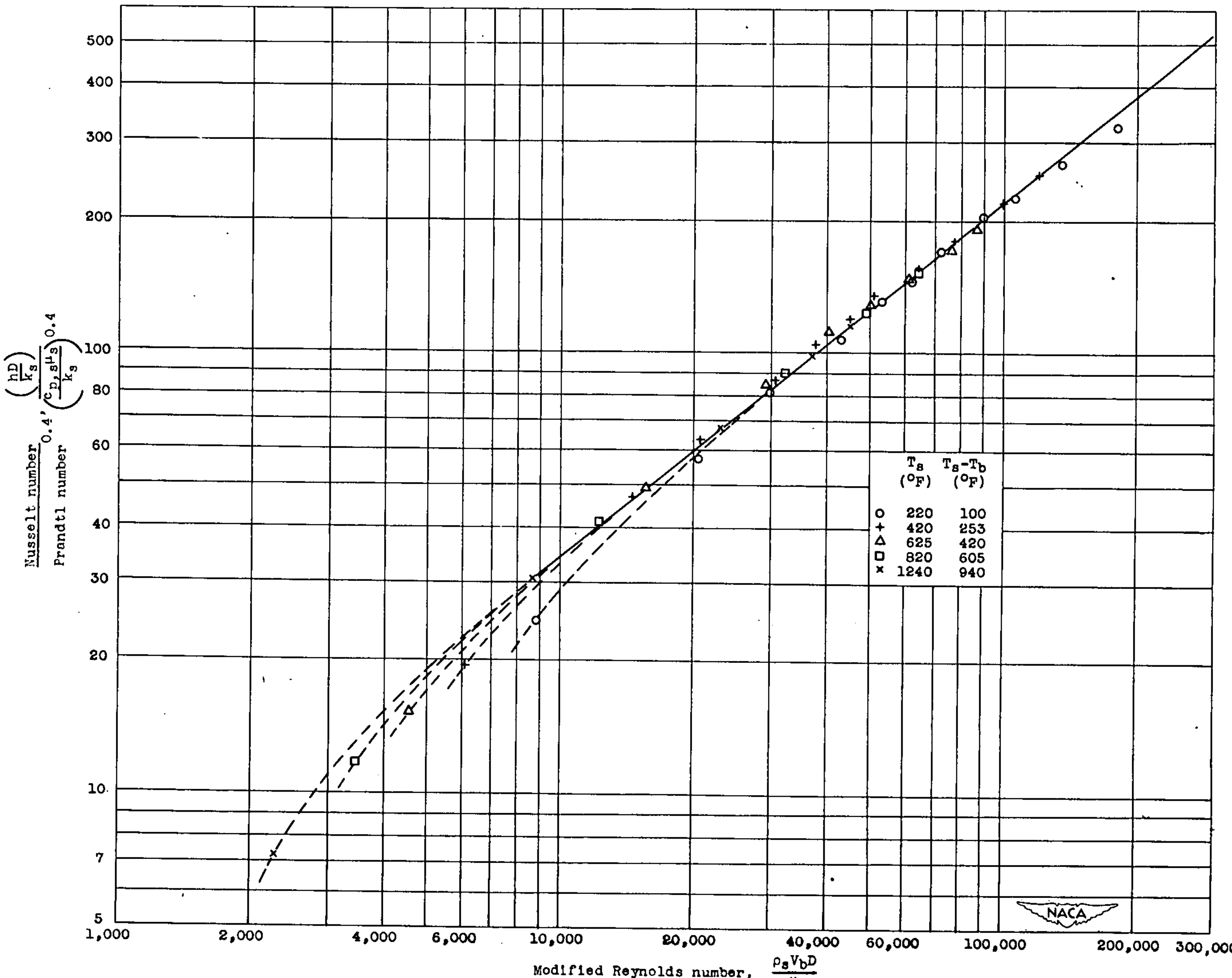

Figure 12. - Correlation of heat-transfer coefficients using a modified Reynolds number. Physical properties of the air evaluated at the average inside-tube-wall temperature T_s.

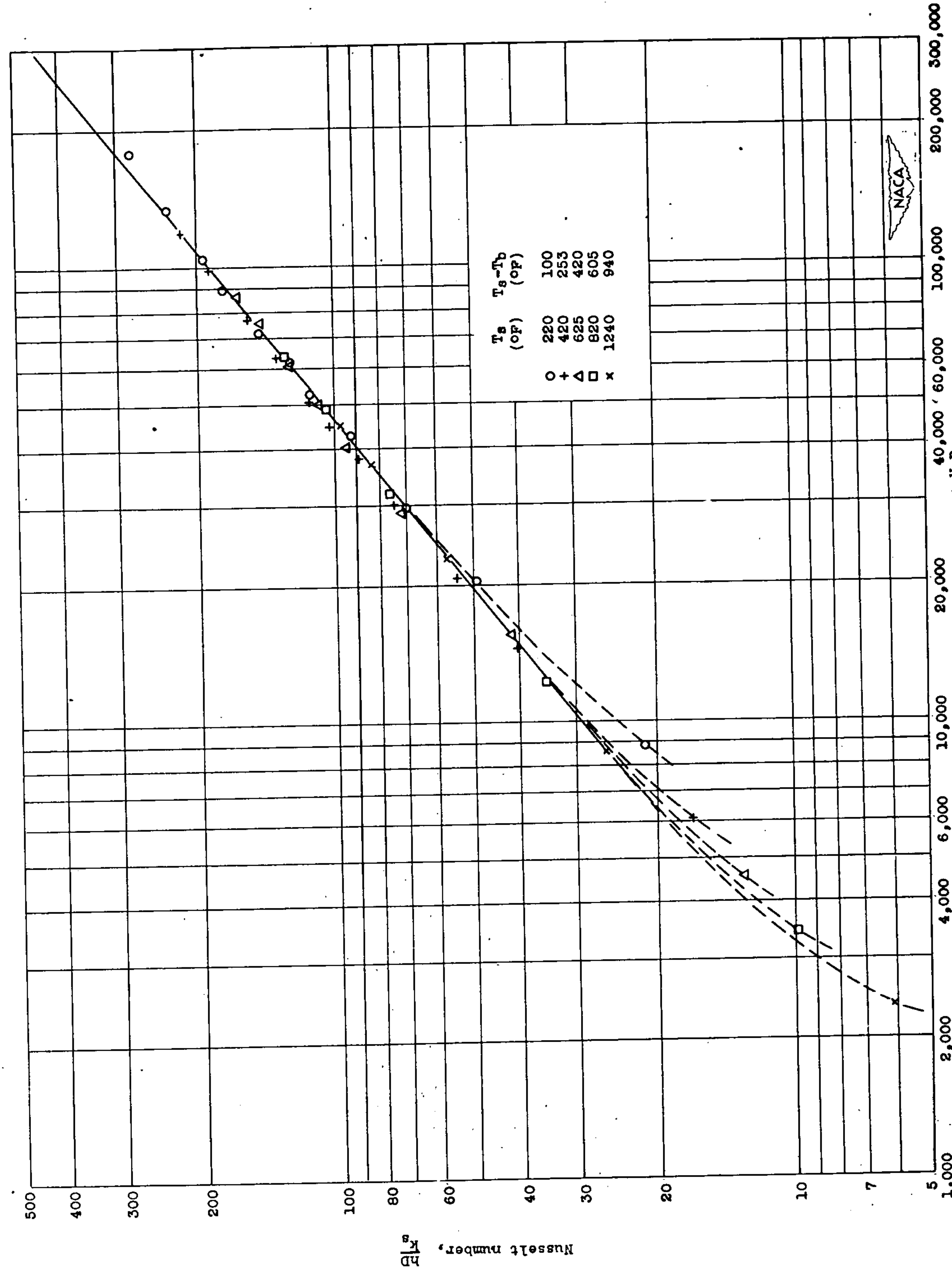

Figure 13. - Correlation of heat-transfer coefficients using a modified Reynolds number and neglecting Prandtl number. Physical properties of the air evaluated at the average inside-tube-wall temperature T_s.

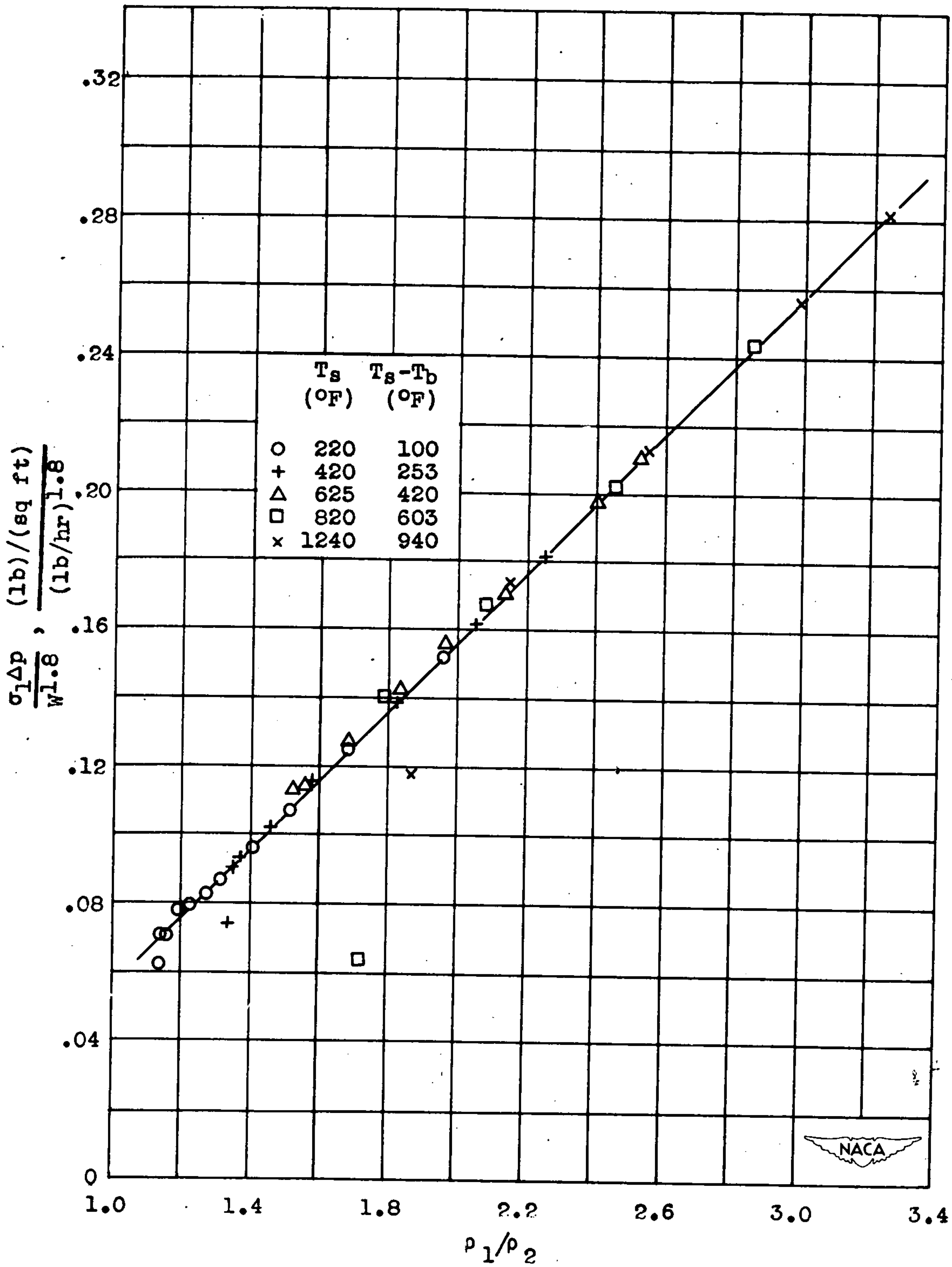

Figure 14. - Correlation of the measured static-pressure drops across the heater tube for various air-flow rates and heat-transfer rates.

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By Leroy V. Humble, Warren H. Lowdermilk
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SUMMARY

A preliminary heat-transfer investigation was conducted with air flowing through an electrically heated Inconel tube with a rounded entrance, an inside diameter of 0.402 inch, and a length of 24 inches over a range of Reynolds numbers up to 250,000 and a range of average inside-tube-wall temperatures up to 1240° F. The corresponding maximum local tube-wall temperatures ranged up to 1500° F.

Correlation of the heat-transfer data by the conventional method wherein physical properties of the air were evaluated at the average bulk temperature resulted in a reduction of Nusselt number of about 25 percent for an increase in average surface temperature from 220° to 1240° F at constant Reynolds number. A good correlation of the data for the entire temperature range was obtained, however, when the physical properties of the air were evaluated at the average surface temperature and the Reynolds number was modified by substituting the product of air density evaluated at the average surface temperature and velocity evaluated at the average air bulk temperature for the conventional mass flow per unit cross-sectional area.

Static pressure drops were obtained for all operating conditions and were satisfactorily correlated with air flow rate and the ratio of entrance-to-exit density.

INTRODUCTION

Most of the fundamental data available on forced convection heat transfer between surfaces and fluids have been obtained at relatively low temperatures and heat-flux densities and do not extend into the range of high temperatures and fluxes that are of interest in many current engineering applications. The accuracy of extrapolation of the existing data to high-temperature and high-flux conditions is doubtful because of the possible existence of extreme temperature and velocity gradients in the fluid film, changes in physical properties of the fluid, and radiation effects that may become important for the case of surface-to-gas heat transfer; changes in boiling phenomenon will markedly affect surface-to-liquid heat transfer.

Accordingly, an experimental investigation has been instituted at the NACA Cleveland laboratory to obtain surface-to-fluid heat transfer and associated pressure-drop information over a range of temperatures and heat fluxes. As part of the general program, an investigation is being conducted with air flowing through an electrically heated Inconel tube having an inside diameter of 0.402 inch and an effective heat-transfer length of 24 inches. Experiments have been conducted with a rounded tube entrance over ranges of Reynolds numbers up to 250,000, average tube-wall temperatures up to 1240° F with corresponding maximum local tube-wall temperatures up to 1500° F, heat fluxes up to 100,000 Btu per hour per square foot, and tube-exit Mach numbers up to 1.0.

The results of these preliminary investigations, which were obtained during the fall of 1947, are reported herein. Average heat-transfer coefficients are correlated in accordance with the familiar Nusselt relation and with a modification thereof; pressure drops are correlated with air-flow rate and density changes. The correlation of these preliminary data, although very satisfactory, will be subject to further investigation over a greater range of conditions.

APPARATUS

A schematic diagram of the heater tube and the associated equipment used in this investigation is shown in figure 1.

Heater Tube

The details of the heater tube are shown in figure 2. The test section consists of an Inconel tube having an inside diameter